

High Speed Reciprocating Compressors – The Importance of Interactive Modeling

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ABSTRACT

Cost-effective, reliable operation of reciprocating compressors requires a balance between pulsation control and performance. With high-speed units, acoustic excitation frequencies more severely overlap the acoustic and mechanical response frequencies of the passages and the piping system. To achieve good cylinder performance and pulsation control, the designer must more than ever understand the capabilities and limitations of the tools used for design.

Internal gas passage acoustics, cylinder-to-cylinder and cylinder-to-piping interactions, and non-linear flow losses can have a significant effect on the accuracy of the design predictions. Industry standard digital models typically solve the linear frequency domain piping response equations coupled to a piston face velocity source. Typically, these models are non-interactive with the piping and, therefore, the predicted cylinder performance does not accurately reflect the acoustics of the cylinder passages or the attached piping. Analog models (which also solve the linear acoustics of the piping system) do, in a limited fashion, allow interaction of the piping and the cylinder. However, this interaction is controlled by the very nature of the analog electrical components and, for high-speed units, the analog can under predict valve and pulsation effects.

The Gas Machinery Research Council (GMRC) has funded an effort to develop a digital design system with full interaction between cylinder and piping components, enhanced to meet the demanding requirements of today's high speed compressor packages. This paper presents the initial results of this effort. This enhanced model accurately reflects the effect of the internal gas passages and piping acoustics on the performance of the cylinder. A comparison is made between field data, predictions from the enhanced digital model, predictions from a non-interactive cylinder and piping digital model, and predictions from an analog simulation. Differences in the capabilities of the various design tools investigated are summarized.

INTRODUCTION

The use of high-speed reciprocating compressors for natural gas transmission, field extraction, and process plant applications is an increasing trend. Case studies have clearly shown that the successful start-up and operation of high-speed units is fundamentally related to the quality of the original design and is very sensitive to off-design operation.

Recent field analyses indicate that high-speed units can experience diminished cylinder performance due to higher than expected valve losses and pulsation levels. These high pulsations consequently contribute to the higher than anticipated vibration levels found on a number of high-speed reciprocating compressors.

Some design tools applied today lack the ability to account for full interaction of the piping and the cylinder. A fully interactive digital design system that addresses the broad spectrum of design and performance issues related to high-speed compressor installations has been developed as part of the GMRC research program.

MOTIVATION

The need, in part, for a fully interactive digital design system has arisen from the inability to accurately model valve losses and to determine the effects of pulsation and dynamic pressure losses on performance.

Valve Loss Calculations

The ability to accurately predict cylinder performance during the design process must include accurate models of valve losses. From physical principles, valve losses are proportional to the square of the gas velocity through the valve. When gas velocities are low, a simple linear model of valve losses has proved adequate for design purposes. In high-speed machines, however, the valve area and internal passage cross-section is much smaller in proportion to cylinder volume than in low-speed machines, resulting in higher gas velocities

through the valve on the suction and discharge strokes. The valve and passage losses for high-speed units can, therefore, depart significantly from existing model predictions.

The valve and passage losses in high-speed machines become critical influences on cylinder performance. Experience from field investigations has shown that valve and passage losses can exceed expected values by 10 % or more on high-speed units.

The Effect of Pulsation on Performance

The effect of pulsation on performance in high-speed units can be significant. Because compressor excitation frequencies tend to overlap piping resonances more extensively in high-speed installations, excessive pulsations can more severely affect compressor and piping performance than in low-speed installations with the same capacity. Worse, many high-speed installations use variable speed drives for capacity control, which means the design must be able to deal with excitation frequencies covering a broader band than the typical low-speed installation. As pulsation levels increase behind the valve, the cylinder pressure required to open the valve must rise. This, in turn, increases the horsepower consumed by the cylinder during compression. Field experience confirms these high pulsation effects on high-speed unit performance.

Dynamic Pressure Drop

Dynamic pressure drop refers to the calculation of instantaneous pressure drop at each point through the crank rotation, integrated over a cylinder cycle, to compute the horsepower cost. This horsepower must be developed by the compressor cylinder and provided by the prime mover. If the instantaneous velocity across a square law restriction is much higher than the mean velocity, then the cycle integrated horsepower costs will be much higher than the horsepower cost estimated by mean (average) flow estimates. Field analyses have documented instances where dynamic ΔP effects have incurred hundreds of excess horsepower and, in some cases, were so high as to inhibit full load operations. Using existing design tools, we can estimate this horsepower term by measuring the flow across the various key locations individually. This horsepower, however, needs to be more fully reflected in the pressure-volume diagram and in the design process. New design tools are needed to automate the dynamic pressure loss calculations throughout a compressor-piping network. The resulting cumulative dynamic resistances would then be used to update the pressure-volume diagram and the horsepower and capacity calculations.

Unloader Volumes

Unloaders can significantly affect pulsation spectrums by driving cylinder-to-cylinder resonances and exciting local piping resonances. Increases in pulsation levels increase losses and thereby decrease cylinder performance. Accurate prediction of passage responses, which will tend to be more excited with the cylinder ends unevenly loaded, is required and represents a significant area of concern for high-speed units. Another area of design concern for high-speed units is the modeling of internal gas passages. Representing the passage as a one-dimensional acoustic space rather than a two-dimensional acoustic space may be important. The modeling tool under development will allow for more advanced geometries to be included. The importance of this will require field validation for a variety of cylinder passage geometries.

Case Studies

A reciprocating compressor unit operating at a fixed speed of 1000 rpm was recently field-tested to determine the cause of poor cylinder performance. The compressor unit under investigation is equipped with volume pockets located on the head end of each cylinder. Figure 1 presents a performance summary.

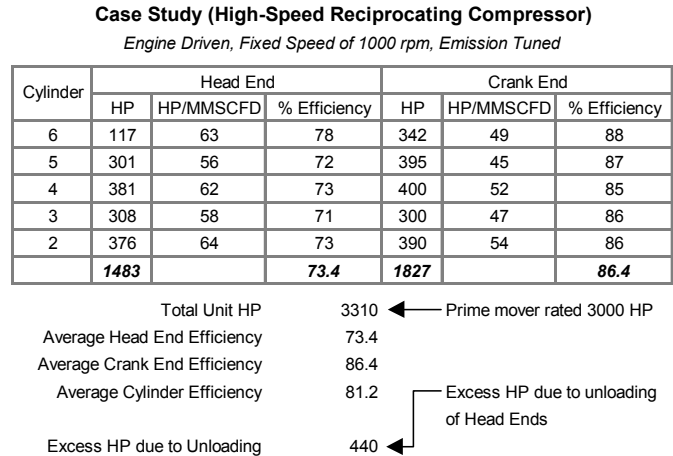


Figure 1. Performance Summary

Head end and crank end performance, as measured using either IHP/MMSCFD or theoretical efficiency, is significantly different. For the unloaded head end, the average efficiency is 73.4%. For the fully loaded crank end, the average efficiency is 86.4%. Total unit horsepower is 3310, which exceeds the engine’s rated horsepower of 3000. Unit capacity is reduced due to a loss in cylinder performance. Although the unit was designed with a variable speed drive, emission

considerations fixed the speed at 1000 rpm and unloading was the only available method of achieving capacity control.

The problem appears to be pulsation driven by the uneven loading on the cylinder. Significant flow losses across the pulsation bottle were noted for simultaneous measurements between the cylinder, nozzle and piping transducers. We also suspect that proper accounting of the flow losses (dynamic ΔP in the bottles, the valves and across the cylinders) combined with accurate pulsation and performance predictions could have avoided this situation.

In another case, on-site testing of a reciprocating compressor unit operating in a natural gas transmission service was conducted to determine the cause of poor cylinder performance. This unit is equipped with two large bore compressor cylinders with multiple pockets on each cylinder end.

As shown in Figure 2, initial performance data showed that the total losses were approximately 55% of the indicated horsepower with the compressor operating on the bottom load step at rated speed.

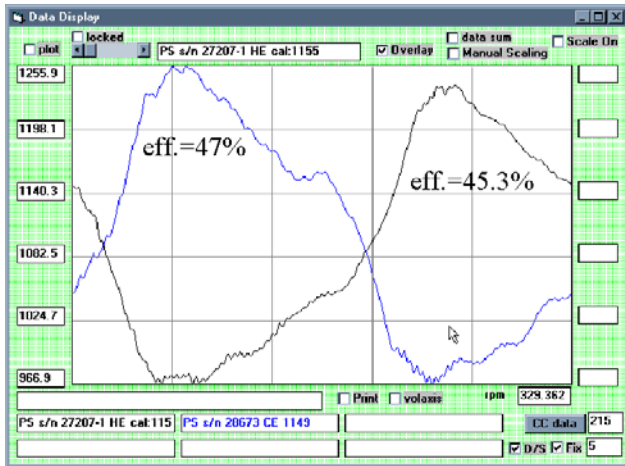


Figure 2. Cylinder Performance before Modifications

The compressor cylinders were modeled digitally to determine the root cause of excessive horsepower losses. The simulation showed that the high horsepower losses could only be reproduced with a significant restriction downstream of the valves and that the mean flow losses were small compared to dynamic losses. This is characteristic of dynamic pressure drop effects.

The pulsation bottle design was subsequently re-evaluated to determine the dynamic pressure drop through the internal choke tubes. As shown in Figure 3, each bottle was designed with an internal baffle and choke tube

assembly in order to minimize the acoustic shaking forces. The end of the choke tube was capped and holes were added to reduce the resonant response.

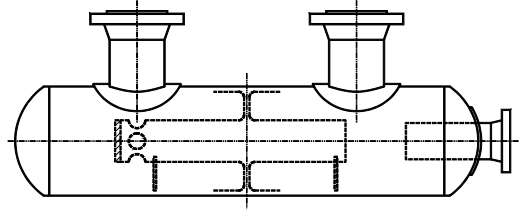


Figure 3. Pulsation Bottle Design

A digital simulation of the compressor cylinders was constructed and combined with an analog simulation of the bottles and the attached piping. The dynamic flows and resulting dynamic pressure drop through the holes at the end of each internal choke tube were estimated. As shown in Figure 4, the mean flow through the restriction was estimated at 2400 acfm with a peak flow of approximately 10,000 acfm. Based on our predictions, some flow reversal is expected. The predicted dynamic pressure drop over one cycle is also shown. As seen, the average value is 17 psi, with a peak value of roughly 50 psi. Note that the pressure drop predicted using the mean flow and the same loss coefficient is only 2.8 psi. The predicted instantaneous horsepower over one cycle is shown as 600 horsepower, where the value based on mean flows and mean pressure drop is only 30 horsepower.

In order to reduce the high horsepower losses, it was recommended that the end cap on each choke tube be removed and that the length of each choke tube be reduced. Once the choke tubes were modified, a follow-up field test was conducted to determine the impact on horsepower. As shown in Figure 5, the head end efficiency increased from 45% to 65% and the crank end efficiency increased from 47% to 71%. This resulted in a total gain of 440 horsepower per cylinder compared to the predicted gain of 600 horsepower.

Based on fuel flow measurements, the unit under investigation still did not meet the design point guarantees. Performance decreased as the operating speed increased and as the units were unloaded. Subsequent measurements indicated that pulsation levels (including nozzle resonances) were driving the losses. Figure 6 presents the lost horsepower versus the sum of the pulsation levels at the discharge and suction flanges. As seen, total combined pulsation levels would have to be less than 40 psi to achieve reasonable cylinder performance.

As with the previous case, we suspect that proper accounting of the flow losses, elimination of the strong cylinder to cylinder interaction, combined with accurate pulsation and performance predictions could have avoided this situation.

These case studies illustrate several of the key factors motivating the development of an enhanced design tool. Specifically, significant cylinder performance losses arising from cylinder to cylinder, and cylinder to piping interaction.

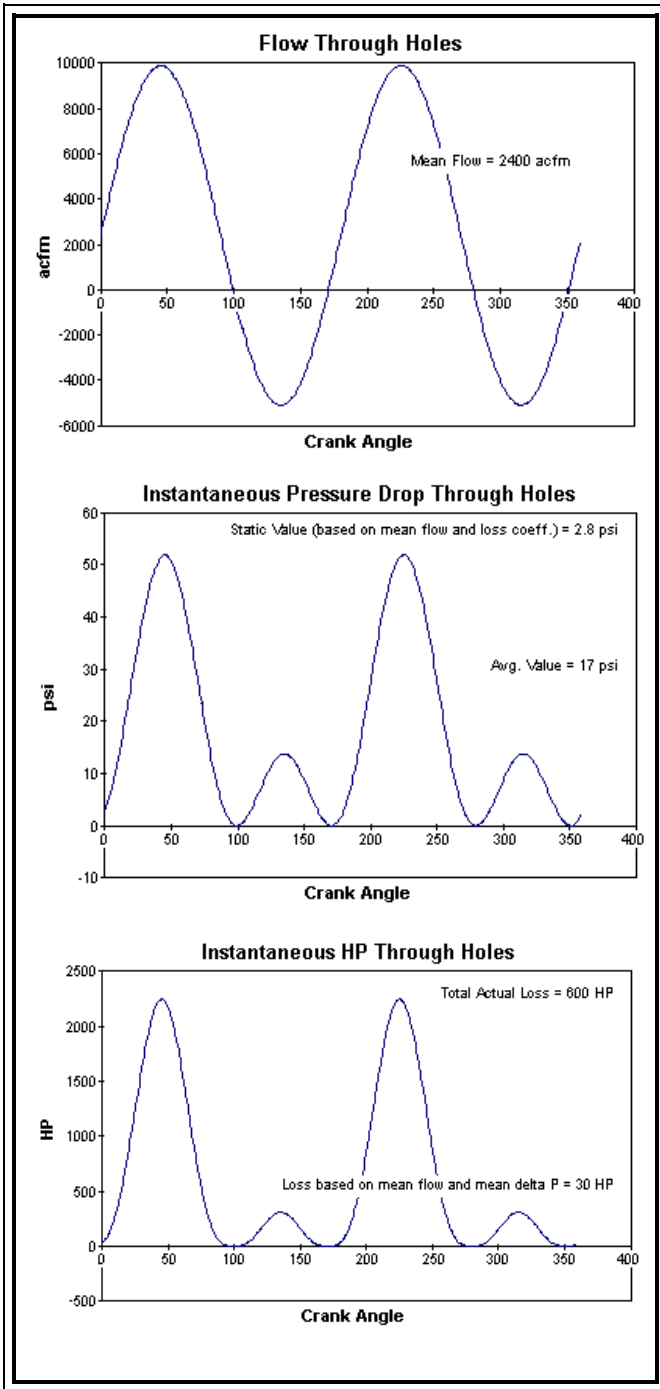


Figure 4. Horsepower Loss through Choke Tube

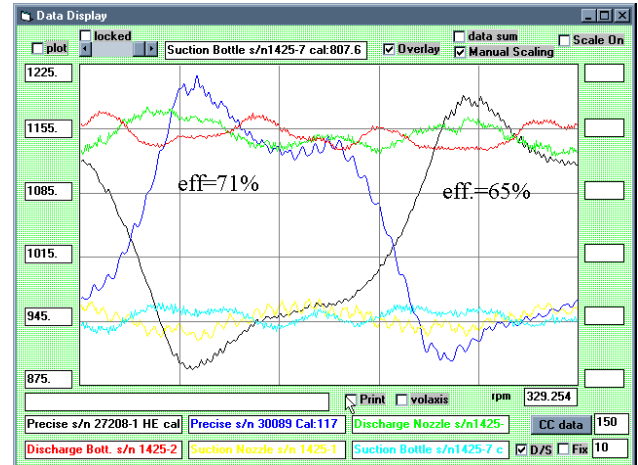


Figure 5. Cylinder Performance after Modifications

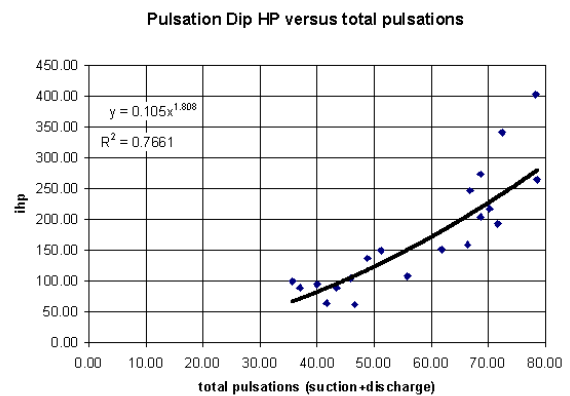


Figure 6. Lost Horsepower vs. Total Pulsations

DIGITAL MODEL DESCRIPTION

For the enhanced, high-speed digital acoustic model, we have chosen an Acoustic Wave Element (AWE) procedure¹ to describe the passage and piping acoustics. Using this approach, the piping is modeled as an assembly of acoustic elements (volumes, junctions, branches, and straight sections). This procedure readily translates the acoustic “line diagram” used by the designer to a mathematical description of the piping acoustics.

These elements are assembled into a global matrix describing the acoustic characteristics of the system. The assembled global acoustic matrix is defined such that:

$$[G(f)][wp(f)] = [\dot{m}(f)] \quad (1)$$

Where:

$$\begin{aligned} [G] &= \text{Assembled Global Acoustic Matrix} \\ [wp] &= \text{Column Vector of Acoustic Wave Amplitudes} \\ [\dot{m}] &= \text{Column Vector of Cylinder End Mass Flows} \end{aligned}$$

Equation 1 can be solved for the acoustic wave pair amplitudes which yield the response dynamic pressures as:

$$\{P(f)\} = [Z(f)] * [\dot{m}(f)] \quad (2)$$

Where:

$$\begin{aligned} [Z] &= \text{Global Impedance Matrix (psi/(lbs/sec))} \\ &= [G]^{-1} \end{aligned}$$

Once the cylinder mass flows have been determined, Equation 2 is then used to predict the dynamic pressures at all locations simultaneously. Note that the global acoustic matrix is largely sparse (99% zeroes) and is developed for each frequency. The sparseness of the acoustic matrix allows for significant reduction in memory and storage (disk) requirements, which must be considered given that this matrix is assembled and then inverted for each frequency under consideration.

For a non-interactive piping/cylinder model, the spectral components of the piston face velocity flows are appropriately located in the mass flow rate column vector of Equation 2. Pre-multiplying by the global impedance matrix yields the predicted piping response dynamic pressures at all locations in the model, assuming that these pressures do not influence the valve flow out of the cylinders.

For a piping/cylinder model where the piping response effects the valve flows, the solution becomes significantly more challenging. An explicit formulation for the piping/cylinder interaction is presented below.

The driving point impedance and all cross coupling terms are obtained using Equation 1 with unit flows (in the frequency domain) sequentially applied at the cylinder valve locations. This is expressed as:

$$\{Z_{ij}(f)\} = [Z(f)] * \{I(f)\} \quad (3)$$

The subscripts i and j denote the source and response locations. For a two cylinder double acting compressor, Equation 3 is applied with unit sources imposed individually at each cylinder end and with the response computed simultaneously at all cylinder ends. Thus for a two-cylinder double-acting compressor, a driving point impedance matrix of dimension [4x4] is generated. This matrix has the dimensions of psi/(lbs/sec) and defines the pressures at the cylinder end locations per unit flow at each cylinder end. Again, this matrix is computed over all desired frequencies. The diagonal entries of the driving point impedance matrix represent the pressure generated at that cylinder end per unit flow (lbs/sec) at that location. The off-diagonal terms define the cross coupling or cylinder end to cylinder end interaction as a function of frequency. In this formulation, the designer can see directly the frequency domain interaction between ends and cylinders from the proposed piping design without completing the cylinder/piping response.

With the driving point impedance function computed, the dynamic pressures behind the valves can be computed as:

$$\{P_{valve}(f)\} = [Z_{dp}(f)] * \{\dot{m}(f)\} \quad (4)$$

Where:

$$\begin{aligned} [Z_{dp}(f)] &= \text{Driving point impedance functions} \\ \{\dot{m}(f)\} &= \text{Spectral components of the cylinder end valve mass flow rates (lbs/sec) generated by the cylinder model} \end{aligned}$$

The cylinder pressures and flow rates are generated using a fundamental thermodynamic model of the cylinder. This model incorporates a dynamic valve model with flow squared losses, and can be enhanced to any level of sophistication including valve stiction, non-linear spring forces or other effects. Since the valve flow is a function of the difference between the cylinder pressure and the pressure behind the valve, estimation of the valve mass flow rates is an iterative process. The valve flow is initially estimated assuming constant pressure behind the valves; the dynamic pressure behind the valve is then computed using Equation 4. With the updated valve dynamic pressures, the cylinder pressures and flow rates are re-computed. This iterative process allows the frequency domain solution of the cylinder piping to be combined with the time domain solution of the cylinder pressures. When the process is complete, the full cylinder-to-cylinder and cylinder-to-piping interactive

valve mass flows have been computed. The dynamic pressures throughout the piping system are then calculated in the frequency domain using Equation 2.

FEATURES OF INTERACTIVE DIGITAL MODEL

The pressure-volume plots predicted using the interactive digital model are realistic based on the horsepower calculations and appearance. The distortion of the pressure-volume plots shown in Figure 7 is indicative of high cylinder valve losses. Figure 8 shows pressure-volume plots of the same cylinder with increased valve lift. Notice that the card is not as distorted and the pulsation effects are clearly visible. The digital cylinder model provides performance summaries, predicted flows based on the thermodynamic model, and overall cylinder efficiencies.

Using the enhanced digital modeling technique, we are now able to model the stepless Hydrocom unloader, a recent Hoerbiger technology that appears to be emerging as a viable approach for load control. This technology^{2,3} uses a hydraulic actuator timed with a key phase, to hold the valve open past bottom dead center to control flow. This approach results in gas flowing into the cylinder during the suction event, and back out the cylinder during a portion of the normal compression stroke. As a result, the suction valve experiences complete flow reversal. This flow reversal has the potential to significantly alter the pulsation on the suction side piping. The new cylinder model predicts the effects of the Hydrocom and the associated flow reversal. Figure 9 shows the pressure-volume plots predicted at 100% load, and 50% load. Notice that the IHP per MMSCF for the 100% loaded case is 4.0 while for the unloaded case, it is predicted to be 4.3. In this case, our model predicts an increase in IHP per MMSCF of roughly 7%.

A comparison between the enhanced digital model and the piston face flow model shows that the enhanced digital model more accurately predicts the spectral components of the predicted flow rates. The use of piston face velocity based valve flows can significantly alter the spectral components of the predicted flow rates, and thereby affect the predicted pulsation levels throughout the attached piping. Figure 10a presents the field measured and predicted cylinder pressure where the valve flow is approximated by a piston face model. Note that there are minimal losses across the suction and discharge events. The suction and discharge flow components for this case are shown in Figure 10b. Figure 10c presents the measured and predicted cylinder pressure using the enhanced digital model. Note that both cases result in the same cylinder flows. Figure 10d presents the corresponding valve flows for figure 10c.

A ratio of the spectral components, shown in Figure 11, indicates that the predicted spectral components of the valve flows are significantly different for the two modeling techniques. The idealized valve flow model (piston face flow) under predicts the spectral components on suction side by as much as 25% to 50% per order. On discharge, where the valve loss is less, the under prediction is between 5% and 30%.

The enhanced digital model explicitly describes the cylinder to piping interactions prior to computing the pulsation levels. The driving point impedance and all cross coupling terms are derived from the piping model prior to combining them with the cylinder flows. This explicit description should encourage the designer to detect potential resonances and strong cylinder to cylinder interactions in the piping system and avoid problems of the type described in the case summaries. Figure 12 presents the driving point impedance looking into the discharge piping for a simple two-cylinder system. Each cylinder discharges into a primary bottle and then a common secondary bottle. The two cylinders are acoustically isolated by the volume-choke-volume filter design. The peak in the suction side driving point impedance, as seen in Figure 12, is at a level of roughly 20 psi/lbs/sec. The resonant frequency in the function is described by the gas passage-nozzle length. As expected, the predicted cross coupling between cylinders is very small, less than 0.5 psi/lbs/sec on suction (Figure 13). We have designed the interactive digital model to allow many of the acoustic functions necessary to predict the cylinder performance and flows to be measured from a traditional analog analysis. In this fashion, the designer can transition or combine technique based on the needs of the client.

Acoustic damping in flowing complex piping systems has long been an area of concern. The analog design approach uses inductor coils whose values set the resonant amplitudes. Correlation with field tests for low speed units has established these values. Typically, lightly damped piping is chosen close to the cylinder and in stub lengths. Further away from the cylinder, increased damping levels are used. Several sources of acoustic damping exist in analog design. The two key components are normal dissipation associated with piping runs and that damping associated with pressure drop across junctions, orifices and choke tubes. We plan a series of field test to acquire meaningful values for high-speed applications. At this time we have established the levels in the digital model to match the high Q and low Q values used in an analog study. Figure 14 presents the driving point impedance functions measured from four different piping spans (two high Q and two low Q). Figure 14 presents the same functions as predicted using the digital model and a frequency dependant damping correlation.

The agreement between the two sets of data is good. At this time, we are automating the flow-based calculation of pressure drop and damping allocation into the model. This will also automate the dynamic pressure drop calculations in the piping system.

SUMMARY

The GMRC funded effort to develop an enhanced digital design tool was motivated by the industry’s trend towards high-speed compression, and the concern that many of these units being installed were performing poorly. The enhanced digital model has many features that should improve our ability to design the acoustic filtering and predict cylinder performance. These features include: realistic thermodynamic cylinder models allowing for the accurate prediction of valve loss effects, the ability to predict Hydrocom equipped units and the associated reverse flow, the ability to model gas passages in a more complicated manner, the explicit determination of the cylinder to piping resonances and the cylinder coupling,

and the automated calculation of dynamic pressure drop effects. Future plans include: field correlation efforts to establish passage effects, further damping testing, and general progression towards the ability to predict pulsation levels accurately enough to allow reasonable trade off assessments between performance and vibration.

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2. Steinruck, Peter, Ph.D., “Smart Compressor Valves for Improved Unit Performance.” Published in Compressor Tech, May/June 1999.
3. Mathews, Hans, “User Experience with Actively Controlled Variable Deactivation (Hydrocom System) for Unit Loading.” Presented at the Gas Machinery Conference, Denver, CO, October 1998.

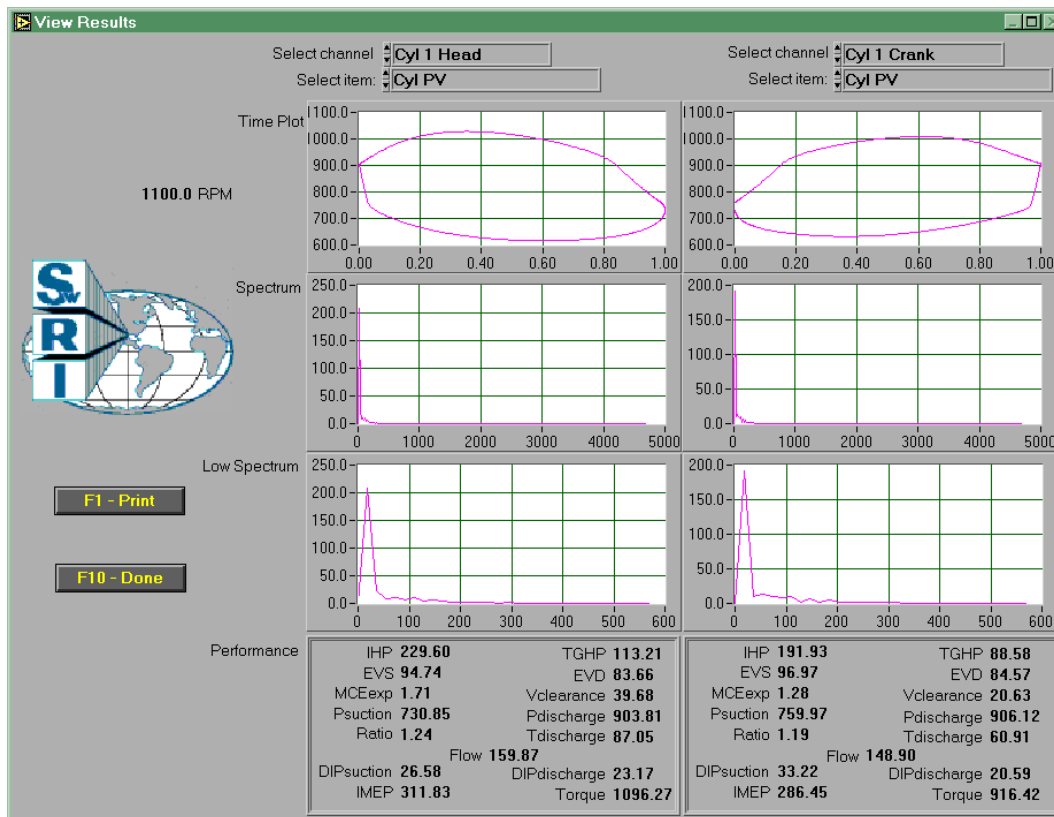


Figure 7. Predicted Pressure-Volume Plot with High Valve Losses

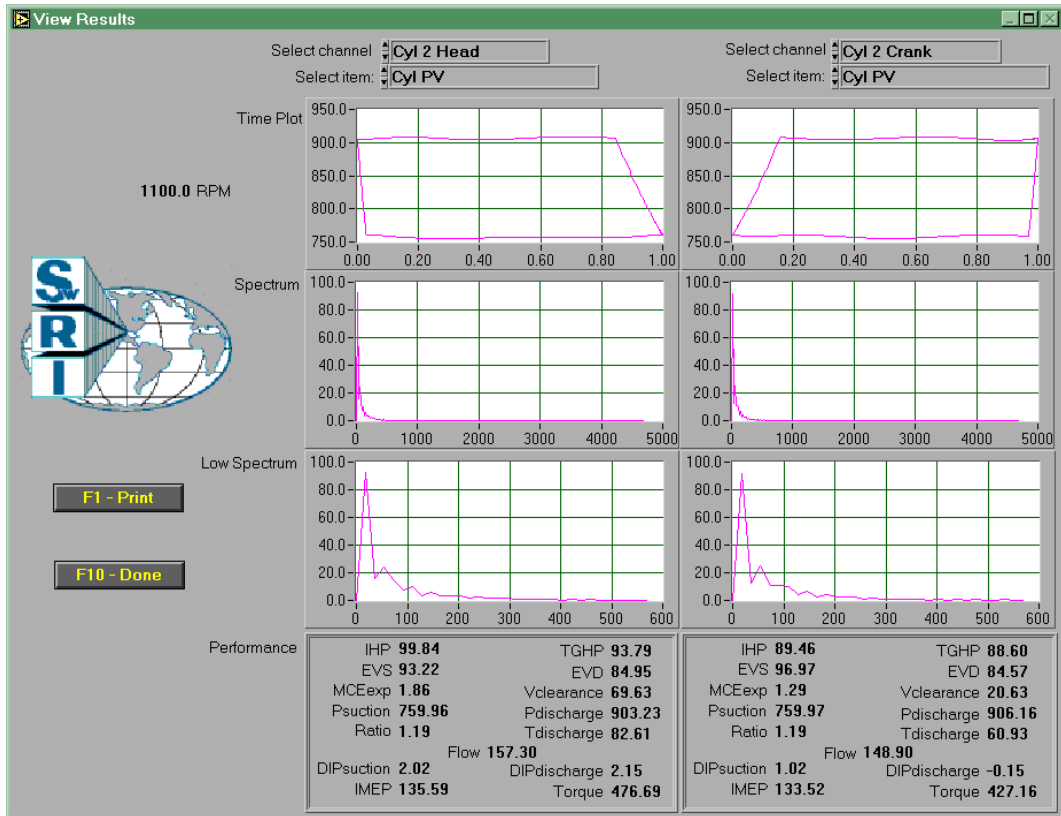


Figure 8. Predicted Pressure-Volume Plot with Increased Valve Lift

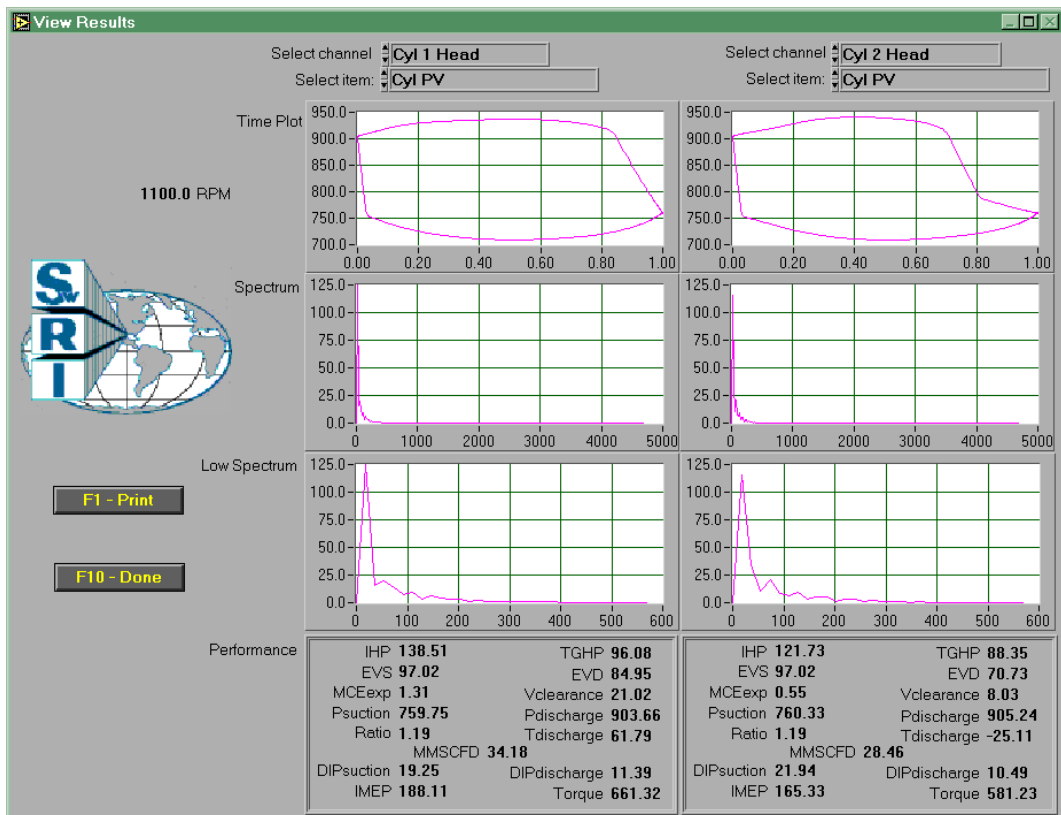


Figure 9. Pressure-Volume Plot with Hydrocom Valves

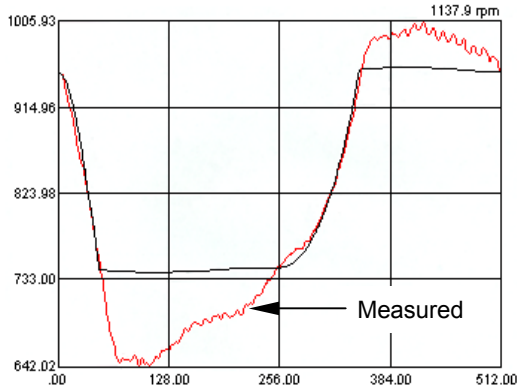


Figure 10a. Cylinder Pressure using Piston Face Velocity Valve Flows

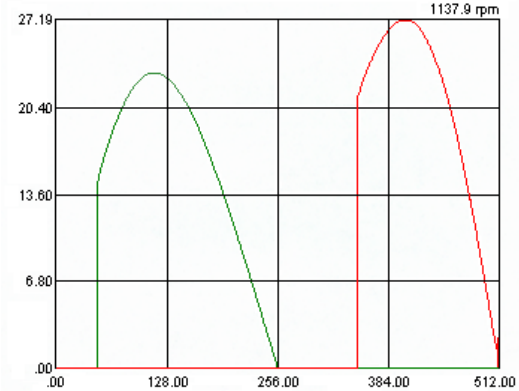


Figure 10b. Suction and Discharge Flows using Piston Face Model

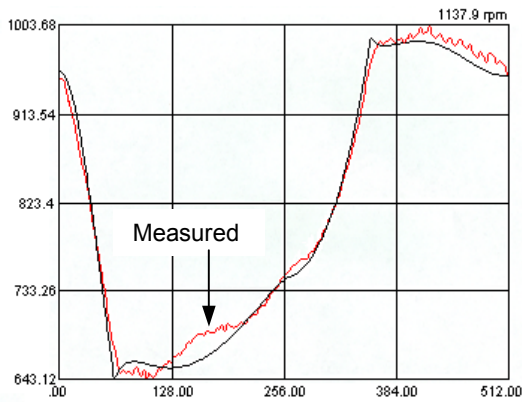


Figure 10c. Cylinder Pressure using Optimized Valve Model

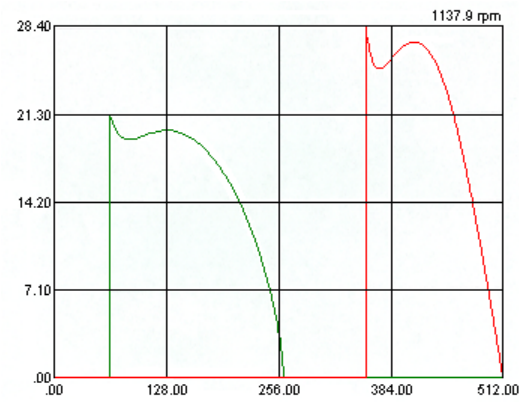


Figure 10d. Suction and Discharge Flows using Optimized Valve Model

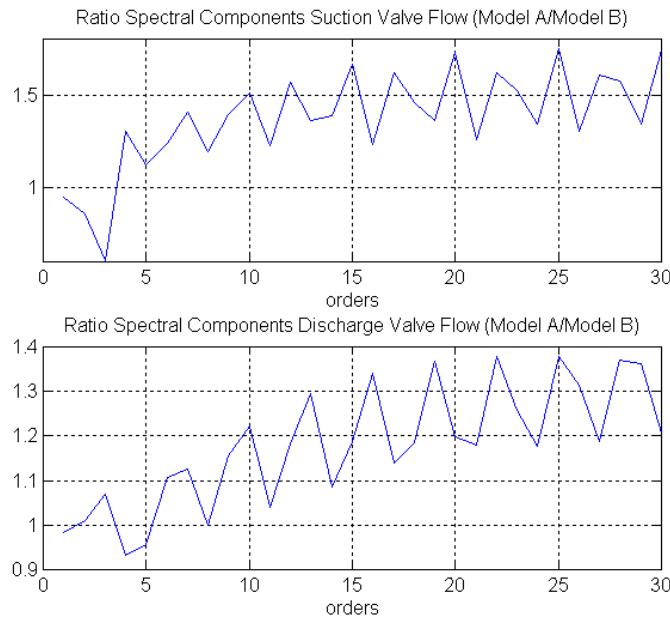


Figure 11. Ratio of Spectral Components

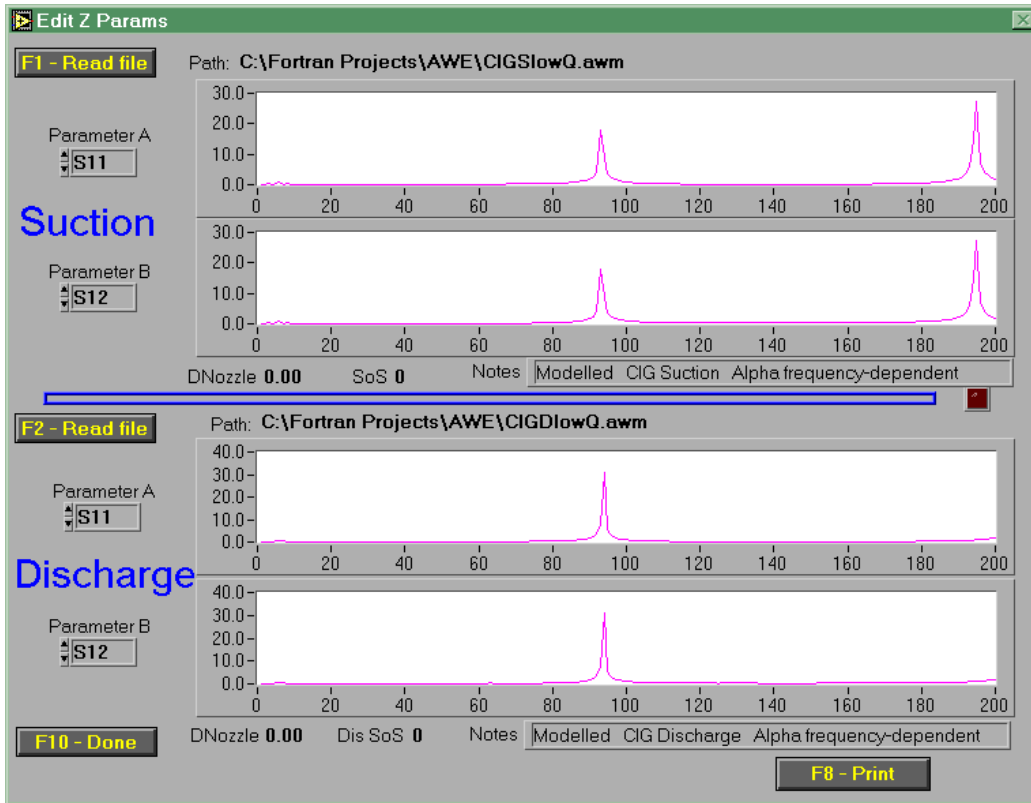


Figure 12. Driving Point Impedance

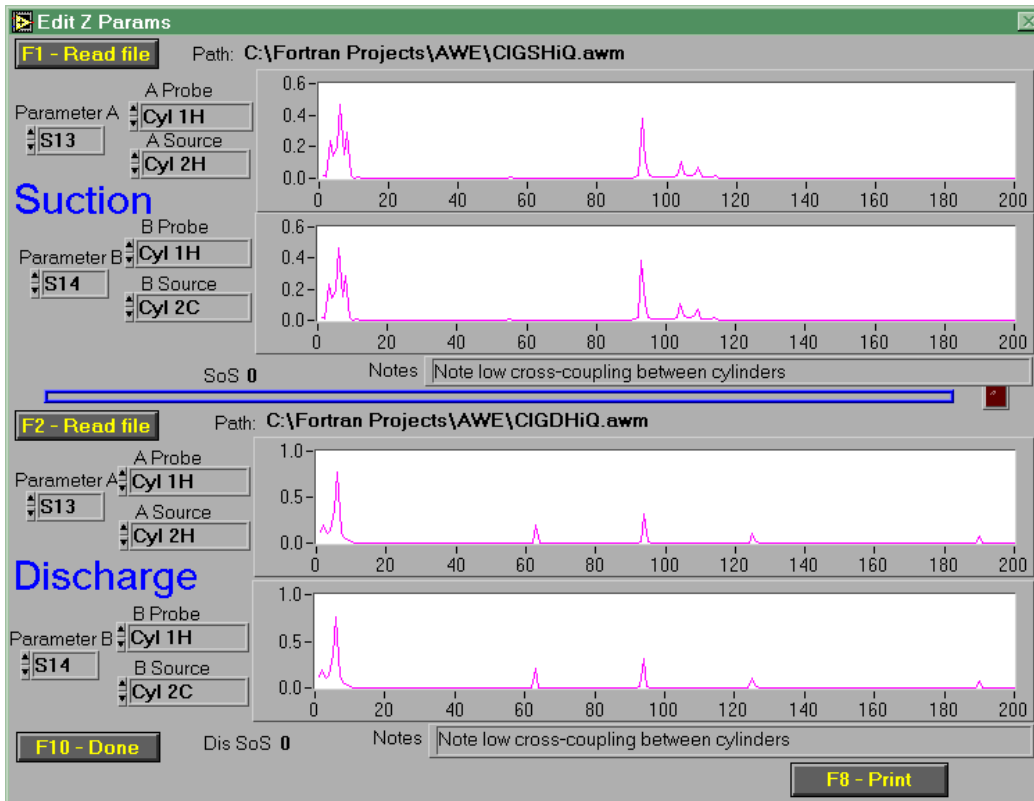


Figure 13. Cross Coupling Terms

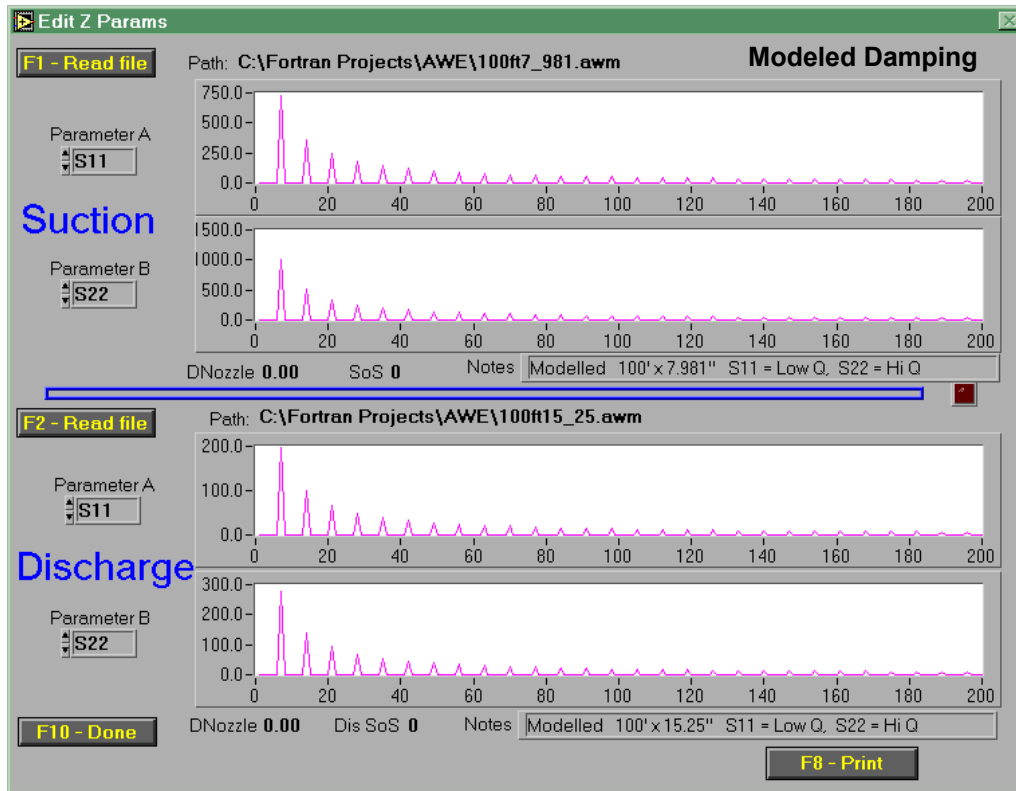


Figure 14. Damping Characteristics